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CONTRIBUTIONS FROM THE JEFFERSON PHYSICAL LABORATORY, HARVARD UNIVERSITY.

THE MEASUREMENT OF HIGH HYDROSTATIC PRESSURE.

I. — *A SIMPLE PRIMARY GAUGE.*

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By P. W. Bridgman.

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Introduction.

The classical work of Amagat on various physical effects of high hydrostatic pressure is practically the only work we have in which the pressure has been accurately measured with a direct reading gauge over any considerable pressure range. Amagat's pressure measurements were made with his *manometre à pistons libres,* which is too well known to need description here. The gauge gives consistent results, and throughout the pressure range the indications are proportional to the pressure. In fact, the accuracy of the pressure measurements is limited only by the accuracy with which the dimensions of the pistons can be measured. With this primary gauge Amagat measured a number of secondary pressure effects, principally the compressibilities of various liquids and gases over a pressure range of about 3000 kgm. per sq. cm. The value of the compressibility found in this way has in turn been used by other experimenters as a means of calibrating whatever secondary gauge they found it convenient to use. It is thus possible to avoid the direct measurement of pressure with a *manometre à pistons libres,* which is in most cases inconvenient, because of the unavoidable leak and the time necessary to make the readings. The care with which the ground surfaces of piston and cylinder must be kept free from grit, and the expense of the instrument, are other objections to its common use.

With increasing experience in methods of reaching high pressures, and increasing excellence of commercial steels, it has been found possible, however, to exceed the pressure limit set by Amagat. Thus

Tammann [1] on one occasion reached 5000 kgm. per sq. cm., and Carnazzi,[2] working with Lussana's [3] apparatus, has also attained 5000 kgm. Both of these observers measured the pressure with a secondary gauge involving directly the compressibility of water as found by Amagat. But because Amagat's data run to only 3000 kgm., the pressure measurements of both Tammann and Carnazzi must be uncertain at these higher pressures. Tammann had to content himself with an extrapolation beyond 3000, and Carnazzi does not give any results beyond 3000.

The purpose of this paper is to provide data which shall enable others to work, if they desire, beyond Amagat's pressure range with a reasonable degree of confidence in the accuracy of the pressure measurements. It seems that the most feasible way of doing this is to determine, under conditions that may be reproduced with accuracy, the variation with pressure of some easily measurable physical property. Compressibility does not seem to be the best secondary property for this purpose, for it cannot be measured with much accuracy conveniently. In this paper, advantage has been taken of a suggestion of de Forest Palmer's,[4] and the variation of the electrical resistance of pure mercury under pressure has been determined. The secondary gauge, involving the variation of the resistance of mercury, has proved itself trustworthy and accurate.

This matter of a secondary standard is discussed in the second part of this paper. The first part is occupied with a discussion of the slightly novel form of gauge with which the fundamental direct measurements of pressure were made. Amagat's *manometre à pistons libres* is not well adapted for high pressures. Amagat himself was accustomed to use it to only 3000 kgm. per sq. cm., and others following him have not been so high; thus Barus found that above 2000 kgm. the leak became troublesome. In this paper a gauge is described with which, by modifying the design and decreasing the dimensions, it has been found possible to reach higher pressures than Amagat, frequently without perceptible leak. In fact the primary gauge proved itself so manageable, and is so simple to construct, that if it were not for the greater convenience of the secondary gauge, the primary gauge could be used directly in any high pressure investigation. This paper gives results that have been obtained with this gauge up to 6800 kgm. per

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1 Tammann, Kristallisieren und Schmelzen, p. 201 (Leipzig, Barth, 1903).
2 Carnazzi, Nuov. Cim., (5), **5**, 180–189 (1903).
3 Lussana, Nuov. Cim., (5), **4**, 371–389 (1902).
4 de Forest Palmer, Amer. Jour. Sci., **6**, 451–454 (1898).

sq. cm. The first part is occupied with a description of the gauge, calculation of the corrections to be applied, and a comparison of two gauges to determine the accuracy and sensitiveness.

Description of the Gauge.

Besides Amagat's [5] manometer, other forms of direct pressure gauge have been used, examples of which are the pressure balance at Stuckrath,[6] and the differential manometer at the National Physical Laboratory at London.[7] Lisell,[8] in his measurement of the pressure coefficient of resistance of wires, used a gauge much like that at Stuckrath. These gauges differ in the manner in which the pressure exerted on the piston is measured. Amagat measures it by measuring with a mercury column the hydrostatic pressure acting on a larger piston which balances the total thrust exerted by the high unknown pressure on a much smaller piston. The thrust is measured at Stuckrath or by Lisell by hanging weights on the piston either directly or with the aid of a lever. At London the action of weights is used to equilibrate the differential effect of the pressure on two pistons of nearly the size. A common feature of all these gauges is the piston fitting accurately in the cylinder, which is subjected to pressure on the inside. The distortion produced by the pressure is, therefore, a compression of the piston, accompanied by a stretching of the cylinder, the resultant effect being to increase the breadth of the crack between piston and cylinder. The leak, therefore, at higher pressures increases because of the increased pressure expelling the liquid and the increased breadth of the crack.

This effect is avoided in the gauge used in this work by subjecting the cylinder in which the piston plays to pressure on the outside as well as on the inside. It is well known that a cylinder subjected to the same pressure externally and internally shrinks to the same extent as a solid cylinder subjected to the same external pressure. By properly decreasing the external pressure on the hollow cylinder, the shrinkage at the inner surface may be made as small as we please, or may be made an expansion. Practically the same result may be obtained by subjecting only a portion of the external surface of the

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[5] Amagat, Ann. de Chim. et Phys., (6), **29**, 544 (1893).

[6] Zeit. f. Instrk., **14**, 307 (1894). Manometer für hohe Druck.

[7] Engineering, **75**, 31 (1903). The Estimation of High Pressures.

[8] Lisell. Om Tryckets Imflytande på det Elektriska Ledningmotståndet hos Metaller, samt en ny Metod att Mäta Höga Tryck. Upsala, 1903 (C. J. Lundström).

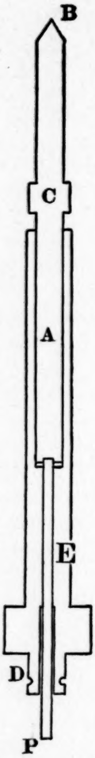


FIGURE 1. The direct reading gauge. P, piston; E, cylinder; A, larger steel rod through which the pressure of the equilibrating weights is transmitted to the piston; B, hardened steel point on which the stirrup carrying the weight pan is hung; C, stop (see Figure 2); D, groove by which the rubber tube containing the viscous mixture of molasses and glycerine is attached.

cylinder to pressure. By suitably changing the area subjected to pressure, the shrinkage of the interior may be controlled. This is the method adopted with the present gauge.

The leak may be further decreased by decreasing as far as possible the dimensions of the piston and cylinder, thus decreasing the circumference of the crack through which leak occurs. Decreasing the size has the additional advantage of making the whole gauge more compact and manageable. In particular, the total thrust becomes small enough to be balanced directly by hanging weights on the free end of the piston. Where the magnitude of the weights is not so great as to make this infeasible, the direct application of weights seems preferable to the usual indirect methods of measuring the thrust. In the gauge adopted in this work, the piston is only $\frac{1}{16}$ in. (0.159 cm.) in diameter, requiring at the maximum pressure of 6800 kgm. an equilibrating weight of about 130 kgm.

The cylinder and piston are shown in Figure 1. In Figure 2 they are shown in place in a large steel block which serves as a reservoir between the gauge and the pressure pump. The dimensions of the important parts are indicated in Figure 3. The thrust on the piston P (Figure 1) is taken by the large cylindrical rod A joined to the piston by a forced fit. A terminates in a hardened point B, on which the weights are hung by a stirrup supporting the scale pan underneath the large steel block. The upper end of the cylinder acts as a guide for the rod A, as does also the attachment screwed onto the top of the cylinder shown in Figure 2. It is essential that fitting here should be accurate, so that the small piston may move freely in a vertical line without danger of any bending of the top end when projecting some distance from the cylinder.

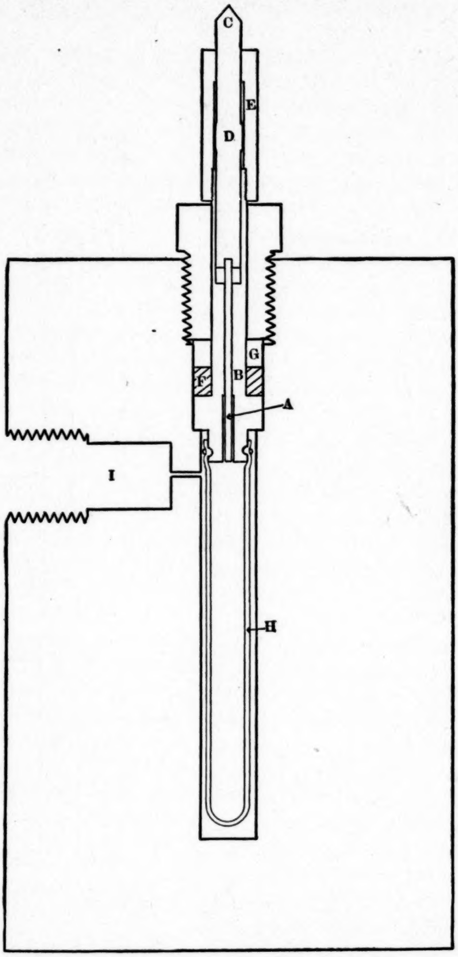


FIGURE 2. A, piston; B, cylinder; C, hardened steel point, on which the equilibrating weights are hung; D, stop, preventing too long a stroke of the piston either up or down. In this stop is placed the rod by which the rotary motion is imparted to the piston to increase sensitiveness. E, guide to insure the upper part of the piston moving rigidly in a straight line. F, rubber packing. G, steel washer, retaining the rubber packing. H, easily collapsible rubber tube, containing the viscous mixture of molasses and glycerine. I, connection to the high pressure pump. The thin mixture of water and glycerine transmitting the pressure is injected through this hole, acts on the outside of the rubber tube H, and so transmits the pressure to the piston A.

The enlargement C, on the rod A, serves as a stop at either end of the stroke, which in this case was $\frac{1}{2}$ in. (1.3 cm.). The piston was made at least $\frac{1}{2}$ in. (1.3 cm.) longer than the hole in the cylinder in which it fits, so that at no part of the stroke is any part of the hole empty. This insures the constancy of the crack through which leak occurs, and ought to increase the accuracy of the results. To diminish friction between piston and cylinder the piston was kept in slow rotary motion through 30° by a rod inserted in a hole in the enlargement C. The rod was driven by a small motor.

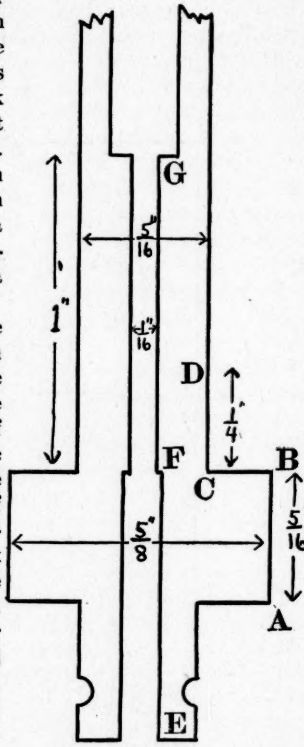


FIGURE 3. Detail, giving the dimensions of the cylinder.

The purpose of the shoulder at the bottom of the cylinder will be plain on an inspection of Figure 2. The disposition of packing, shown by the shading, is one that has proved itself serviceable in other high pressure work. It is obvious from the figure that the pressure on the outside of the cylinder mentioned above as preventing the enlargement of the crack between cylinder and piston is the pressure exerted by this packing. The portion of the cylinder over which it acts may obviously be varied by varying the quantity of packing. With dimensions of cylinder, etc., shown above, $\frac{1}{4}$ in. (0.64 cm.) thickness of packing proved satisfactory.

To go into this question of packing at any length would be beyond the scope of this paper. Neither can any description be given here of the apparatus with which the pressure was produced. Briefly, pressure was produced by a small piston pushed by hydrostatic pressure on a larger piston. Pressure was transmitted to various parts of the apparatus by heavy steel tubing. It is hoped that methods of producing high pressures may be made the subject of another paper.

The cylinder (E, Figure 1) was turned in a lathe from a rod of

about 1.25 per cent carbon tool steel. The drilling of the hole in which the piston moves demanded care. This was drilled first with a drill about 0.002 in. (0.05 mm.) under $\frac{1}{16}$ in. (1.59 mm.), and then enlarged to full size with a two-lipped $\frac{1}{16}$ in. twist drill. The hole made in this manner proved round, uniform, and satisfactory in every particular. It is a matter of common experience that a two-lipped twist drill hugs the hole very tightly when used as a following drill. For this reason, care is necessary not to push the drill too hard, as otherwise the sharp cutting corners are quickly blunted. After turning and drilling, the cylinder was hardened in water, and the temper drawn below a blue. Drawing the temper is a necessary precaution in the interests of safety, as glass hard steel proved itself very treacherous. The cylinder is so small that with the exercise of a little care in heating and quenching it is not distorted appreciably by the hardening.

The piston was a piece of $\frac{1}{16}$ in. (1.59 mm.) "Crescent" drill rod, hardened in oil, the temper not being drawn further. This drill rod was found to be remarkably round and uniform in diameter, variations of so much as 0.0001 in. (0.0025 mm.) being rare from end to end of the same length. Different pieces, however, of nominally the same size rod may differ by 0.0005 in. (0.0125 mm.) in diameter. It was merely necessary, then, to select from several lengths of drill rod a piece fitting the hole in the cylinder. No grinding whatever was necessary, either on the cylinder or the piston, except rubbing with the finest emery paper to remove the film of oxide after hardening. In fact, it is the salvation of this device that no grinding is necessary, accurate grinding of a piston so small as $\frac{1}{16}$ in. being out of the question, to say nothing of the $\frac{1}{16}$ in. hole in the cylinder. Because of its slenderness, considerable care is necessary in hardening the piston without warping. Several attempts were usually necessary before a perfectly straight piston was obtained. This, however, is a matter of no consequence, because a piston can be made in a few minutes. The writer has himself made two cylinders and pistons complete in one day.

Leak was reduced to a very low value by using a liquid of great viscosity to transmit pressure to the piston. A mixture of molasses and glycerine proved suitable. The viscosity can be given any desired value by boiling away enough water from the molasses before adding the glycerine. Besides increasing the viscosity, the glycerine serves the useful purpose of preventing the molasses from drying where it leaks out between piston and cylinder. The liquid used to transmit pressure from the high pressure pump to the gauge was a mixture of two parts glycerine to one part water. This was prevented

from coming into contact with the molasses and glycerine by enclosing the latter in an easily collapsible rubber tube, closed at the lower end, and at the upper end tied over the mouth of the cylinder, as shown in Figure 2.

Molasses was the liquid used by Amagat in his manometer. A heavy mineral oil, such as Barus used in a gauge of Amagat's type, was found to be unsuitable for high pressure work, because it freezes at room temperature under pressure. One grade of heavy oil tried in this experiment froze at 20° under a pressure of 4500 kgm. Presumably vaseline and such soft solids become unsuitable for the same reason, although this point was not tested. For the same reason the glycerine transmitting pressure from the pump had to be diluted with water. The ease with which glycerine subcools, and the difficulty of getting it pure, made any exact determinations impossible; but it was found that commercially pure glycerine was very apt to solidify at 6000 kgm. and 20°.

Corrections to be applied to the Absolute Gauge

In spite of the simplicity of this gauge, and the directness with which it carries the measurement of pressure back to the fundamental definition, there are two corrections which must be applied in practical use. These corrections are both so small, however, that neither need be determined with much accuracy.

The first correction is introduced by the slow leak, and is in amount equal to the frictional force of the escaping liquid on the piston. The equilibrating force must balance both the hydrostatic pressure on the end of the piston and this frictional force. The effect of the correction, therefore, is to increase slightly the effective area of the piston. If we assume that both cylinder and piston are perfectly cylindrical, and that the crack between them is so narrow that the friction exerted by the escaping liquid is equally divided between cylinder and piston, then we easily see by writing down the equations of steady motion of the escaping liquid that the friction increases the effective diameter of the piston to the mean of the diameter of the piston and cylinder. It appears from the equations that this correction is independent of both the rapidity of leak and pressure. This is usually determined by measuring the diameter of the piston directly, and the diameter of the hole in the cylinder by some such indirect method as weighing the quantity of mercury required to fill it. The dimensions of the gauge used here were so small, however, that direct measurement of even the piston could not be made with the desired percentage accuracy,

and accordingly the effective diameter was determined in another way, to be described later.

The second correction is a correction for the distortion of the gauge under pressure, and increases in percentage value directly with the pressure. This correction, of course, varies with the type of gauge, but in the types of gauge described above, and the pressure gauge employed, the correction is practically negligible. A rough calculation showed that at 3000 kgm. the correction in Amagat's manometer is about $\frac{1}{10}$ per cent. Since, however, it was desired in this work to reach an accuracy of $\frac{1}{10}$ per cent, and since the pressure range is 6800 kgm., some approximate evaluation seemed desirable.

No easy experimental method of determining this correction suggested itself, so recourse was had to a calculation, using the theory of elasticity. This was done only as a last resort, because of the doubtful accuracy of the mathematical theory at these pressures, and of the fact that the solution obtained is only an approximation, instead of a rigorous mathematical solution. In fact, the general problem involved has not been solved mathematically, and even if it could be, its application here would be doubtful, because slight irregularities in either cylinder or piston would destroy the ideal boundary conditions of the mathematical problem. In spite of all these objections, however, the magnitude of the approximate correction turned out to be so slight, $\frac{1}{10}$ per cent, that the calculated value can probably be applied with a fair degree of confidence.

The facts used in the following calculation are taken from the most elementary parts of the theory of elasticity, and may be found stated in any book under the calculation of the strains produced in a cylinder by external or internal hydrostatic pressure. It will be noticed that the correction for distortion found below includes the effect of the friction of the escaping liquid.

The strain in the piston can be broken up into two components. The first is that due to the longitudinal compression of the piston by the hydrostatic pressure at one end and the equilibrating weights at the other, and is uniform throughout the piston. The radius increases from this effect by the amount

$$\Delta r = \frac{3\kappa - 2\mu}{18\mu\kappa} \times r \times P,$$

where P is pressure in kgm. per sq. cm., κ the compressibility modulus, and μ the shear modulus. These elastic constants vary only slightly in different grades of steel. If we assume as average values that

$$\mu = 7.8 \times 10^5 \text{ kgm./cm.}^2,$$

we find that
$$\kappa = 15 \times 10^5 \text{ kgm./cm.}^2,$$
$$\Delta r = 1.4 \times 10^{-7} \times r \times P.$$

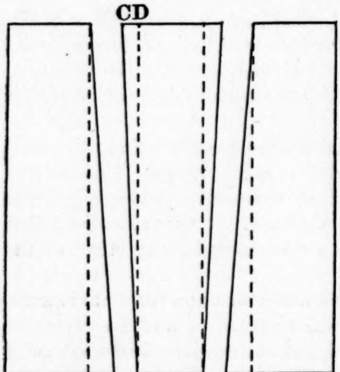
The second component part of the strain is that due to the pressure of the escaping liquid over the curved surface of the piston. Here an approximation must be introduced, for the determination of the strain in a cylinder under a given system of normal stresses on the curved surface seems to be a mathematical problem not yet solved, while in this case the problem is additionally complicated because the stress system is not given but depends in turn on the strain. The approximation made is the assumption that the radial displacement at any point is proportional to the normal pressure at that point, and is the same as that in an infinite cylinder subjected to the same pressure over its entire length. This assumption is probably fairly close to the truth where the extent of the cylinder exposed to the pressure is long compared with the radius, and the pressure varies gradually from point to point, as is the case here.

The piston then assumes under the external pressure the form of a frustum of cone, as is shown in Figure 4. It will appear in the following that it is absolutely immaterial whether the generating lines of the frustrum of the cone into which the piston has been deformed are straight, as drawn in Figure 4, or not. The displacement at any point due to the external pressure is, on the above assumptions,

$$\Delta r = -\frac{4\mu + 3\kappa}{18\mu\kappa} \times r \times p$$
$$= -3.5 \times 10^{-7} \times r \times p.$$

p increases along the piston from its full value, P, at the inner end, E, to zero at the outer end, C. Now by adding these two components of strain, we find that the total radial displacement of the piston consists of a shrinkage of $2.1 \times 10^{-7} \times r \times P$ at the inner end, and a swelling of $1.4 \times 10^{-7} \times r \times P$ at the outer end.

CD



A B E

FIGURE 4. Exaggerated effect of the pressure in distorting the cylinder and piston.

The strain in the cylinder is more difficult to compute because of the uncertainty in the external boundary conditions introduced by the packing. Upon the portion of surface DCB (Figure 3) there is a normal pressure exerted by the packing, equal to 1.32 of the internal hydrostatic pressure. On BAEF there is the normal hydrostatic pressure, and from F to G the same distribution of pressure as on the piston, decreasing from the full value at F to zero at G. The maximum radial displacement due to external pressure may be taken as somewhat less than that from a pressure equal to 1.32 P over the entire external surface, because of the supporting action of the part AB, which is subjected to P only, and of the part beyond D, on which there is no pressure. An upper limit to the distortion is probably set by the distortion of an infinite cylinder subjected to 1.32 P on the outside, and P on the inside. This gives

$$\Delta r = \left(- \frac{0.32\, a^2}{2\,\mu\,(a^2 - b^2)} + \frac{4\,\mu + 3\,\kappa}{18\,\mu\kappa} \cdot \frac{b^2 - 1.32\, a^2}{a^2 - b^2} \right) \times b \times P$$

$$= -\,6.9 \times 10^{-7} \times b \times P$$

where a is the external radius, $\frac{5}{16}$ in. (0.79 cm.), and b the internal radius, $\frac{1}{16}$ in. (0.16 cm.). A value probably nearer the truth is found by assuming for the effective external pressure 1.16 P, i. e., a mean between the maximum and the pressure on AB. This gives

$$\Delta r = \left(- \frac{0.16\, a^2}{2\,\mu(a^2 - b^2)} + \frac{4\,\mu + 3\,\kappa}{18\,\mu\kappa} \cdot \frac{b^2 - 1.16\, a^2}{a^2 - b^2} \right) b \times P$$

$$= -\,5.3 \times 10^{-7} \times b \times P$$

and this value will be used in this computation. This represents the maximum radial displacement of the cylinder, which occurs at the inner end; at the outer end there is no pressure either external or internal, and the displacement will be assumed to vanish. Throughout the length of the cylinder the displacement at the inner surface will be assumed proportional to the internal pressure at that point, although the approximation is not so good here as for the piston.

From these displacements of piston and cylinder it is now required to correct for the change in the effective area of the piston. We do this by considering the equilibrium of the escaping liquid. The piston and cylinder each exert on the liquid approximately the same frictional force (F). Furthermore, the cylinder exerts on the escaping liquid a pressure P_1, which is the negative of the component in the direction of the axis of the pressure of the liquid in the crack on the cylinder. P_1 corresponds, therefore, to the axial component of pres-

sure on a ring of breadth AB (Figure 4). Similarly the piston exerts a pressure P_2 equivalent to that on a ring CD. The free liquid at the inner end exerts P_3 on the ring BE. Since the liquid escapes steadily without acceleration, we have

$$2F + P_2 = P_1 + P_3.$$

The effective force on the piston is $F + P_2$

$$F + P_2 = \frac{P_1 + P_3 + P_2}{2}.$$

We now can calculate P_1 and P_2 without any assumption as to the distribution of pressure in the crack if we assume only that at every point the radial displacement is proportional to the pressure at that point. This gives

$$P_1 = 2\pi R \int_{r_1}^{r_2} pdr,$$

where r_1 is the value of r at the end ABE of the cylinder; and r_2 at the end CD. R is the average of r_1 and r_2. But

$$\begin{aligned} r_2 - r &= Cp, \\ dr &= -Cdp, \end{aligned}$$

$$\begin{aligned} P_1 &= -2\pi CR \int_{P_A}^{P_C} pdp \\ &= 2\pi C \frac{RP^2}{2} = 2\pi \frac{P(r_2 - r_1)}{2} R. \end{aligned}$$

That is, P_1 is equal to the pressure exerted by the total internal pressure P on a ring of half the breadth of AB. Similarly, P_2 is the pressure on a ring of half the breadth of CD. If now we put R equal original radius of piston, and $R + \Delta R$ equal original radius of cylinder,

$$\begin{aligned} AB &= 5.3 \times 10^{-7} \times (R + \Delta R) \times P, \\ CD &= 3.5 \times 10^{-7} \times R \times P, \\ BE &= \Delta R + (2.1 \times 10^{-7} - 5.3 \times 10^{-7}) \times R \times P, \\ &= \Delta R - 3.2 \times 10^{-7} \times R \times P. \end{aligned}$$

Hence, $$F + P_2 = 2\pi R \frac{(2.6 + 1.8 - 3.2)\, 10^{-7} \times R + \Delta R}{2} \times P$$

$$= 2\pi R \left(\frac{\Delta R}{2} + 1.2 \times 10^{-7} \times R\right) \times P.$$

This force, $F + P_2$, acts in addition to the hydrostatic pressure on the inner end of the piston, which is now decreased in radius by $2.1 \times 10^{-7} \times R \times P$. The new effective radius is therefore

$$R + \frac{\Delta R}{2} - (2.1 - 1.2) \times 10^{-7} \times R \times P,$$

as compared with the original effective radius $R + \Delta R/2$. The correction on the area is therefore $2 \times (2.1 - 1.2) \times 10^{-7} \times P$, or 0.018 per cent per 1000 kgm. The correction turns out, as was to be expected, independent of the size of the crack.

If the maximum value given above for the distortion of the cylinder is used, the effective radius will be found to be

$$R + \frac{\Delta R}{2} - 1.7 \times 10^{-7} \times R \times P,$$

which gives a maximum correction of 0.034 per cent per 1000 kgm. per sq. cm. Experimental reasons will be given later for preferring the lower value for the correction. This value, 0.018 per cent per 1000 kgm., was therefore the correction applied in all the subsequent work.

The Gauge in Practical Use.

The first essential in making an actual measurement with this gauge is a knowledge of the effective area of the piston. As has been intimated above, this could not be determined directly because of the smallness of the parts, and an indirect method was therefore adopted. Briefly, this consisted in subjecting simultaneously to the same hydrostatic pressure the small piston and another piston large enough to be measured accurately, and finding the equilibrating weights required on the two pistons. The effective areas are then in the ratio of the equilibrating weights.

The larger piston was ¼ in. (0.635 cm.) in diameter, 2 in. (5.18 cm.) long, ground to fit a reamed ¼ in. hole in a large cylinder of Bessemer steel. As this larger gauge was intended for use only to 1000 kgm., the increased breadth of crack produced by exerting the pressure on the interior only of the cylinder was not great enough to give troublesome leak. Also the correction to the effective cross section due to distortion is small enough to be entirely neglected at 1000 kgm. The diameter of the ¼ in. piston could be measured certainly to one part in 2500 with a Brown and Sharpe micrometer. The hole in the cylinder was not measured by filling with mercury and weighing, or by any such frequently employed device. It was instead carefully tested

against the piston while the latter was in process of being ground to size. The piston was too large to enter the hole except by forcing, when 0.0001 in (0.00025 cm.) larger than the final size. This allowance is probably too much, but still probably not so high as to make the error introduced here in the effective area as much as $\frac{1}{10}$ per cent. This method of measuring the diameter of a hole by testing against plugs of known size is the method used by Brown and Sharpe themselves, and is probably the most accurate that we have, when it is possible to obtain the comparison plugs. The comparison of piston and cylinder was easy in this case because all the work was done in the machine shop of this laboratory.

As preliminary work with this larger gauge, a Bourdon gauge by the Société Genevoise was calibrated to 1000 kgm., and showed a maximum error of 5 kgm. per sq. cm. Various liquids were used to transmit pressure to the $\frac{1}{4}$ in. piston, from vaseline which gave a barely perceptible leak, to a thin mixture of water and glycerine, with which the leak was so rapid that pressure could be maintained only with difficulty. The indications of the gauge, as compared with the Bourdon gauge, proved independent of the rapidity of leak, as they should. In the use of the gauge, sensitiveness was secured as usual, by keeping the piston in continual rotation. Made sensitive in this way, the gauge was very much more sensitive than the Bourdon gauge, responding to about one part in 20,000 at 1000 kgm.

Two high pressure gauges of the type described above were compared with this $\frac{1}{4}$ in. gauge at 1000 kgm. Pressure was kept constant during the comparison by the rise or fall of the $\frac{1}{4}$ in. piston, which had a long enough stroke to accomplish this. As was to be expected, the larger piston proved more sensitive than the smaller ones. The certainty of rise or fall of the small pistons was made greater by observing them with the telescope of a cathetometer. The method of proceeding was to apply a constant weight to the small piston, and then find the two weights on the large piston for which the small piston just began to rise or fall. To accomplish this, the weight on the large piston had to be changed by 0.4 kgm. with a total load of 300 kgm. The mean of these two extreme values gives, therefore, the true equilibrating weight to certainly $\frac{1}{10}$ per cent, and probably much better than this.

From the effective area of either piston found in this way, and the measured diameter, the size of crack between piston and cylinder can be computed. It turned out to be 0.0001 in. (0.00025 cm.) for one gauge, and 0.0003 in. (0.00075 cm.) for the other. This was roughly verified by the more rapid leak shown at higher pressures by the latter

gauge. With the former gauge the leak was almost imperceptible after pressure had been kept at 7000 kgm. for an hour. It is a curious fact that the leak around the more loosely fitting piston was distinctly most rapid at 2000 kgm. The decreased leak at higher pressures may probably be taken as proof of the efficiency of the application of pressure to the outside of the cylinder in decreasing the size of the crack, although there is a slight possibility that the effect is due to increased viscosity of the molasses under pressure.

With this calibration, the critical examination of the behavior of the gauges might have been terminated, because the simplicity of the construction is such as to make improbable any error in their use. As a matter of fact, the indications of the various types of gauge described above have usually been accepted at their face value, without comparing with any other absolute gauge. There were means at hand in the present case, however, of so easily comparing the one gauge with the other that it seemed worth while doing. The method adopted was an indirect one, depending on the secondary mercury gauge described in the second part of this paper. It had been found from a great many preliminary comparisons of different mercury gauges that the indications of the mercury gauge were constant, giving a trustworthy measurement of pressure, if once the calibration with a primary gauge could be effected. More detailed proof of this statement will be found in the second part. The two absolute gauges described above were, therefore, compared at different times against the same mercury gauge, and the two sets of readings compared.

The results of the comparisons are shown in Table I. Gauge I was compared twice with the mercury resistance, and Gauge II once. Each number entered in the table is the mean of two or four readings made at increasing or decreasing pressures. The agreement of the two readings under increasing or decreasing pressure, as also of the readings of Guage I on two separate occasions, was as close as it was possible to make the measurements of change of resistance, and, therefore, only averages have been tabulated. The change of resistance could be read to one part in 3000, at the maximum pressure. The average divergence of the readings of either gauge from the mean is well under $\frac{1}{10}$ per cent. The readings of Gauge II are consistently higher than those of Gauge I, a discrepancy which would point to a slight error in determining the effective area of the pistons. The discrepancies also show a tendency to become larger at the higher pressures. This is probably no fault of the gauges themselves, but may be due to the increased difficulty of making fine adjustments of pressure at the higher values. The method of procedure was to apply a known

weight to the piston, and then vary the pressure until equilibrium was produced. Setting on this equilibrium pressure was made more difficult by the fact that pressure always showed a tendency to fall after an increase, and to rise after a decrease, a fact that may be explained

TABLE I.

COMPARISON OF TWO ABSOLUTE GAUGES AGAINST THE SAME MERCURY GAUGE.

Gauge I.		Gauge II.		$\frac{\Delta R}{R_0}$ from Gauge I at Gauge II Pressures.	Percentage Divergence from Mean.
Pressure kgm./cm.^{2}	$\frac{\Delta R}{R_0}$.	Pressure kgm./cm.^{2}	$\frac{\Delta R}{R_0}$.		
917	0.002862	929	0.002898	0.002897	−0.015
1501	0.004555	1519	0.004605	0.004604	−0.012
2018	0.005960	2043	0.006032	0.006025	−0.05
2602	0.007491	2634	0.007577	0.007572	−0.03
3196	0.008989	3235	0.009095	0.009083	−0.05
3779	0.010390	3825	0.010530	0.010500	−0.10
4233	0.011420	4285	0.011560	0.011530	−0.15
4816	0.012740	4864	0.012840	0.012840	−0.00
5348	0.013860	5414	0.014020	0.013990	−0.10
5932	0.015030	6005	0.015220	0.015180	−0.13
6452	0.016070	6531	0.016290	0.016230	−0.20
6841	0.016820	....	....	....	....

The absolute gauges were not corrected for distortion, as this is not necessary for the comparison.

by thermal effects of compression, but is more probably due to elastic after effects in the containing steel vessels. It may be concluded, therefore, from the agreement of these comparisons, that even if all the error is in the absolute gauge and none in the mercury resistance, that this type of gauge is good to about $\frac{1}{10}$ per cent.

The comparison with mercury gauges also furnished an estimate of

the sensitiveness of the gauge. It was found that throughout the entire pressure range the pistons would respond to differences of pressure that could not be detected by the change of electrical resistance. At 7000 kgm., therefore, the gauges remain sensitive to at least 2 kgm. per sq. cm. The continued sensitiveness of the piston with the crack only 0.0001 in. furnishes an argument against the maximum value set, in the discussion above, on the distortion of the cylinder. For, if we accept the above maximum, we shall find that at 7000 the crack must decrease 0.00018 in., or in this case completely close up. There cannot well be an error of this magnitude in the micrometer measurement of the diameter, and the probable correctness of the average value of the distortion used above is thus increased.

Conclusion.

In this first part of the present paper a description has been given of an absolute gauge, so designed that leak does not become troublesome, at least to 6800 kgm. per sq. cm. The various corrections to be applied have been discussed, and the method by which the dimensions were determined has been described. From a comparison of two gauges of this type with one of another type, the probable accuracy of the gauge is estimated to be at least $\frac{1}{10}$ per cent, and the sensitiveness, 2 kgm. per sq. cm., at 7000 kgm. per sq. cm.